

# **STRESS ANALYSIS OF HIGH SPEED TURBOMACHINE**

A Project Report Submitted for Fulfilment of B. Tech.  
In  
Mechanical Engineering

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Under The Guidance  
of  
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**CERTIFICATE**

This is to certify that the project entitled “**Stress Analysis of High Speed Turbomachine**”, being submitted by Mr SURYAKANTA SWAIN (Roll No.– 110ME0429), is a record of bona fide research carried out by him at our institute, **NATIONAL INSTITUTE OF TECHNOLOGY, ROURKELA**, under my guidance and supervision. The work incorporated in this project has not been, to the best of my knowledge, submitted to any other University or Institute for the award of any degree or diploma.

Place --  
Date--

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I am extremely thankful to **Prof. K. P. Maity**, Head, Department of Mechanical Engineering and **Prof. S. K. Sahoo and Prof. S.C.Mohanty**, Course Coordinator for their help and advice during the course of this work.

I am greatly thankful to all the staff members of the department and all my well-wishers, class mates and friends for their inspiration and help.

Place- Rourkela  
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## **ABSTRACT**

The intent of this thesis is to study stresses in high speed turbo machine. A turbo machine used for current investigation is a small turboexpander which include rotor, compressor and turbine wheel. Rotor is a common component used in variety of applications across the industries. When the rotor is subjected to a very high angular velocity, a large centrifugal force acts on the various element of the rotor. It tries to pull the material out. Due to the high rotational speed not only high stress but also deformation comes into picture. Because of the high rotational speed, components like the thrust collar on the shaft and the compressor operate close to their failure stress. Ductile materials are more prone to fatigue failure than the brittle materials because the stress concentrations can't be compensated by local deformations that occur in case of static loading, leading material to fail. Therefore it is essential to analyse the stress and deformation of all the elements before using it in applications. This project concentrates on analysis of rotor, compressor and turbine wheel of a high speed turbomachinery. Main focus was to find stress and deformation of rotor, blades of turbine wheel and compressor wheel. The analysis has been done using ANSYS software and the solid models are created using SolidWorks. The analysis reveals come of the critical areas in different component of turbo machine. Also this thesis gives the detail steps on how to do analysis of turbo machines arts using ANSYS. The author expects the analysis and conclusion will help the engineering society of the world to analysis, design and fabricate turbomachinery for better reliability and efficiency. Current study can be extend further to analytical analysis of high speed rotor using Finite Element Method.

# **CONTENTS**

Certificate-----	ii
Acknowledgement -----	iii
Abstract-----	iv
Contents-----	v
Nomenclature-----	vi
List of figures-----	vii
List of tables-----	viii
<b>CHAPTER -1 -----</b>	<b>1-5</b>
<b>Introduction</b>	
1.1 -- Use of High Speed Rotor	
1.2 – Working Principle	
1.3 – Application	
1.4 – Present Aim of Work	
<b>CHAPTER-2-----</b>	<b>6-11</b>
<b>Literature Review</b>	
2.1 – History	
2.2 – Recent Development	
2.3 - Organizations Involved In Turboexpander Activities	
2.4 - Analysis of Turboexpander	
<b>CHAPTER – 3-----</b>	<b>12-28</b>
<b>Analysis of Turbomachine</b>	
3.1 – Objective	
3.2 - Steps for the Stress and Deflection Analysis	
3.3 - Stress Analysis of Rotor Shaft	
3.4 - Compressor	
3.5 - Analysis of Turbine Wheel	
<b>CHAPTER – 4-----</b>	<b>29</b>
<b>Result and Discussion</b>	
<b>CHAPTER – 5-----</b>	<b>30</b>
<b>Conclusion and Future Scope</b>	
<b>REFERENCES-----</b>	<b>31-32</b>

## **NOMENCLATURE**

<b><math>\omega</math></b>	–	Angular velocity (rad/s)
<b><math>\rho</math></b>	–	Density of the material (kg/m <sup>3</sup> )
<b><math>g</math></b>	–	Acceleration due to gravity (m/s <sup>2</sup> )
<b><math>E</math></b>	–	Young's modulus (Gpa)
<b><math>m</math></b>	–	Mass of the rotor (kg)
<b><math>\nu</math></b>	–	Poison's ratio (dimensionless)
<b><math>N</math></b>	–	Revolution per minute (rpm)
<b><math>r</math></b>	–	Radius of the disc (metre)
<b><math>n_s</math></b>	–	Specific speed (dimensionless)
<b><math>\varnothing</math></b>	–	Diameter (mm)

## **LIST OF FIGURES**

- Figure 1.1** : Layout of A Turboexpander (Source- Ghosh P, (2002) [1]).
- Figure 1.2** : Schematic of Turboexpander (Source- Ghosh P, (2002) [1]).
- Figure 1.3** : Cross Section View of Turboexpander. (Source –Turboexpander Compressor Technology for Ethylene Plant. General Electric).
- Figure 3.1** : Dimension of Shaft Rotor System (source – Ghosh P,(2002)[1]).
- Figure 3.2** : Solid Model Generation Tree on the Left of Figure.
- Figure 3.3** : Mesh Generation Process.
- Figure 3.4** : Various Methods for Mesh Generation.
- Figure 3.5** : Von-Mises Stress of Shaft (Without Any Fillet) in MPa.
- Figure 3.6** : Von-Mises Stress on Shaft (With 3mm Fillet) in MPa.
- Figure 3.7** : Maximum Stress Vs Fillet Radius in rotor.
- Figure 3.8** : Variation of Stresses and Angular Velocity for Filleted and Non-Filleted Shaft Rotor.
- Figure 3.9** : Displacement Plot of Shaft
- Figure 3.10** : Dimension of the Shaft with collar at Middle.
- Figure 3.11** : Von-Mises Stress Plot of Shaft without fillet
- Figure 3.12** : Von-Mises Stress Plot of Shaft with fillet of radius 3 mm
- Figure 3.13** : Deformation Pattern of the Shaft
- Figure 3.14** : Dimension of the Compressor.
- Figure 3.15** : 3D Model of Compressor.
- Figure 3.16** : Von-Mises Stress Plot of Compressor.
- Figure 3.17** : Deformation Plot of Compressor.
- Figure 3.18** : Dimension of Turbine Wheel.
- Figure 3.19** : Von-Mises Stress Plot of Turbine Wheel.
- Figure 3.20** : Deformed Shape of the Turbine Blades

## **LIST OF TABLES**

**Table 1** : Blade Profile of Turbine (Source – Ghosh, S., (2008) [20]).



## **Introduction**

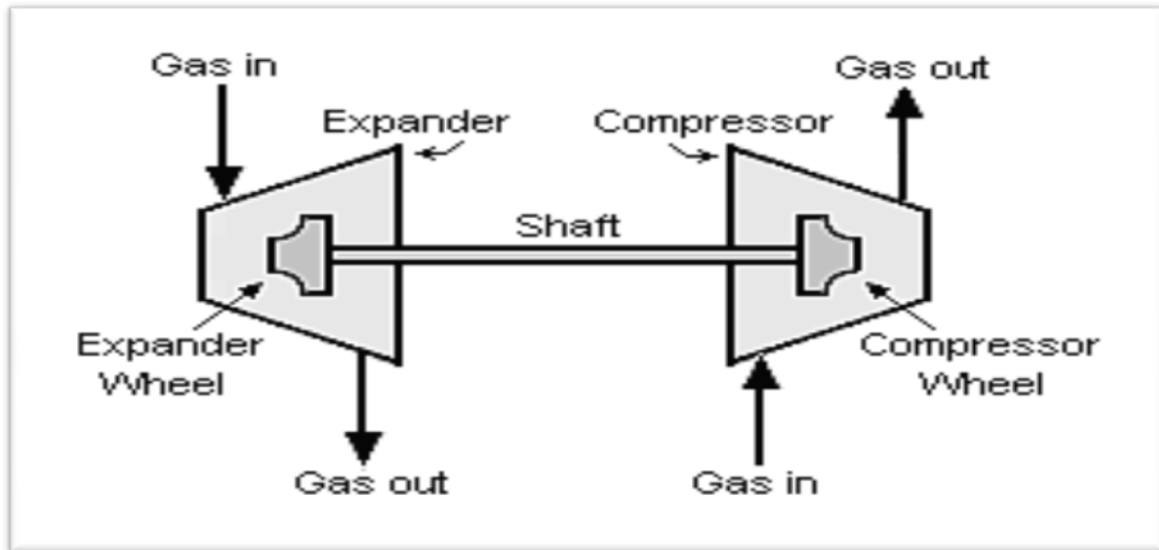
### **1.1 Use of High Speed Rotor**

Rotor is the non-stationary rotating part of machine. Various types of rotor are turbine rotor, electric rotor, helicopter rotor, turboexpander etc. Above all high speed rotation is required in case of Turboexpander for cooling purpose. That's why in this project the rotor shaft of the Turboexpander has been taken for analysis. Turboexpander is also known as expansion turbine. It is having turbine with compressor. The fluid/gas expands through the turbine where it releases its energy to the turbine as rotational energy, this energy is used at the other end to compress another fluid/gas. The expansion is assumed as isentropic process i.e. where entropy remains constant throughout the process i.e. the change in entropy is zero. The gas at exit is at very low pressure and the temperature of the low pressure gas is at a very low temperature i.e.  $-140^{\circ}\text{C}$  or less depending on the various properties of fluid.

Refrigeration industries generally widely use turboexpander in their various processes as the extraction of the gases like ethane and NGLs (Natural Gas Liquids) .

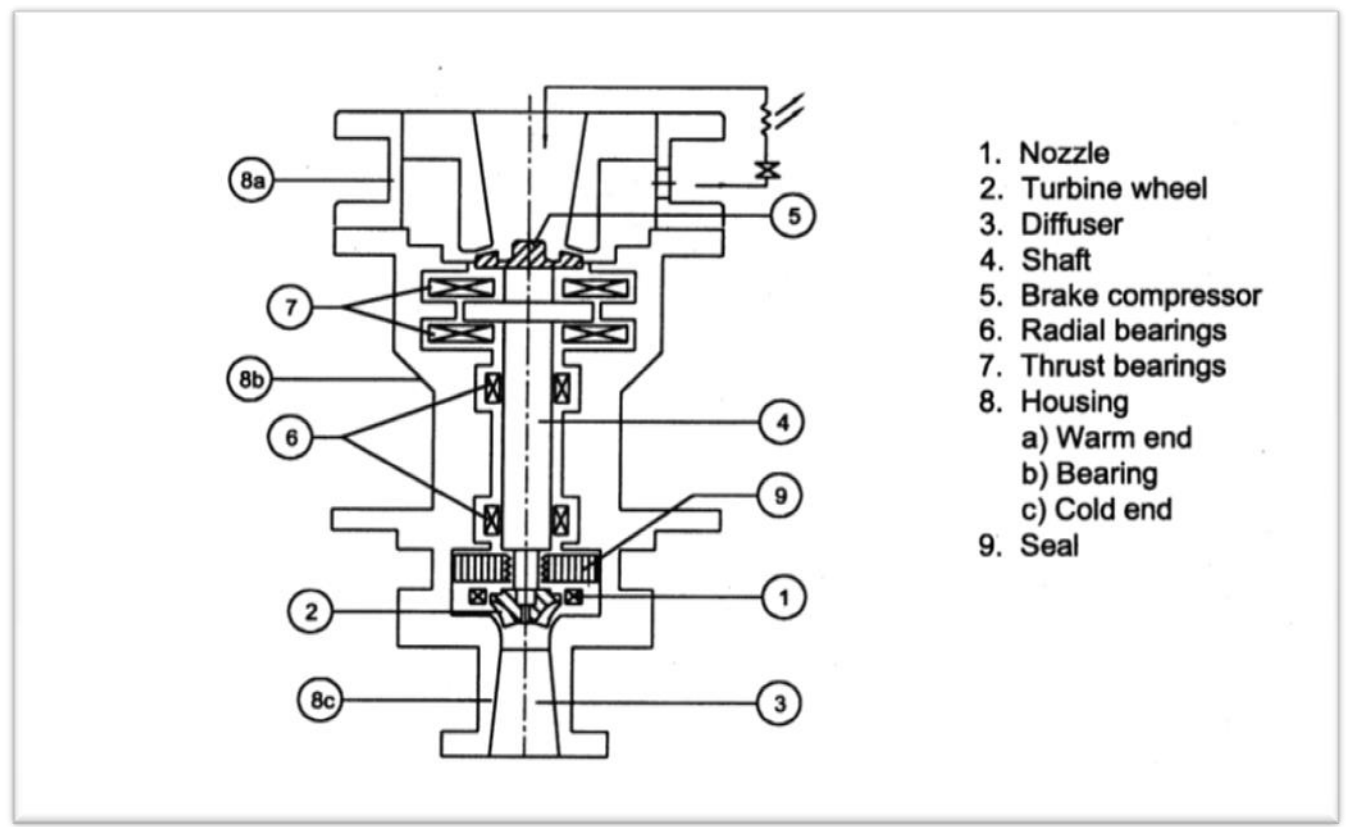
Various parts of a Turboexpander are:

- (1) – turbine wheel
- (2) – compressor wheel
- (3) – shaft
- (4) – thrust bearing
- (5) – journal bearing
- (6) – nozzle ring



**Figure 1.1** - Layout of a Turboexpander (Source- Ghosh P, (2002) [1])

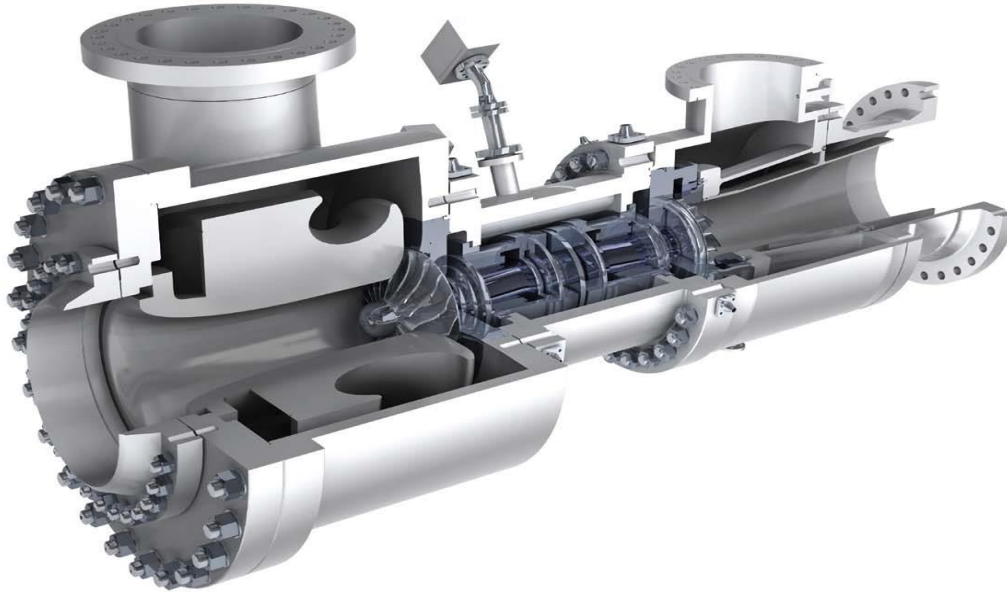
## 1.2 Working Principle



**Figure 1.2** – Schematic of Turboexpander. (Source- Ghosh P, (2002) [1])

The gas at very high pressure enters into the turbine through various piping, into the plenum of the housing and from there into the nozzle. The gas exit from nozzle at very high speed and strikes the impeller radially. This high speed gas imparts torque to the rotor. Generally converging-diverging nozzle is used where the pressure of the gas decreases and velocity of the gas increases at the exit. The pressure energy is transformed into the kinetic energy. From the first law of thermodynamics it is known that energy always remain conserved. As pressure energy converts into the kinetic energy the temperature of the gas decreases. These high speed streams strike against the blade and transmit the forces to the blade and generate torque. The alignment between the nozzle and the blades are made in such a way that the effect due to sudden changes in flow direction and loss of energy can be avoided.

The turbine wheel that is used in Turboexpander is of radial or mixed flow type. The radial or mixed flow turbines are those where the flow enters the wheel radially and exists axially. While larger units are generally shrouded, smaller wheels are open, the turbine housing acting as the shroud. When the high speed flow of gas strikes the blade of the turbine, it loses its velocity. That means the energy of the fluid decreases after the strike. But that energy is being transmitted to the turbine wheel i.e. work is extracted as the gas expands in turbine. This work extracted can be used to compress some other gas at another end, generate electricity etc. Due to the loss of energy the gas at exit is having very low temperature below  $-140^{\circ}\text{C}$  or less. These gases can be used in cryogenic applications like liquefaction of gases. That's why cryogenic industries use high speed turboexpander for their applications.



**Figure 1.3** – cross section view of turboexpander. (Source - Turboexpander-compressor Technology For Ethylene Plant. General Electric)

### **1.3 Application**

Although turboexpanders are very commonly used in low-temperature processes, they are used in many other applications as well. This section discusses one of the low temperature processes as well as some of the other applications.

- (1) – power generation
- (2) -- Extracting hydrocarbon liquids from natural gas
- (3) – Refrigeration system
- (4) – power recovery in fluid catalytic cracker

### **1.4 Present Aim of Work**

Due to high rotational speed materials develops stresses and deformation at various locations and vibrates heavily. Main motive of the thesis is to find critical areas of deformation and stress concentration in high speed rotor, compressor and expansion turbine wheel used in turboexpander. After finding those critical areas of stress concentration from analysis of stress and deformation various modifications in the design has been done. The model, which gives

stress and deformation under the permissible limit of the materials used, is supposed to be the best design. Help of static structural analysis package of ANSYS has been taken to find out critical points, which uses FEM for the analysis of the same. The dimensions of the rotor, compressor and turbine taken for the stress and deformation analysis by ANSYS are shown in **Figure 3.1, Figure 3.10, Figure 3.14, and Figure 3.20** respectively.

### **Literature Review**

#### **2.1 - History**

A detailed survey about the history of turboexpander and its development has been presented by Collins and Cannaday [6] and Sixsmith [7]. Lord Rayleigh was the first to use the concept of using a turbine in refrigeration process in 1988. He explained the critical functionalities of a cryogenic expansion device, i.e. low temperature generation instead of getting mechanical work done. Followed by this suggestion, a number of patents were published on cryogenic expansion turbine. In 1898, Edgar C. Thrupp, a British engineer made a simple liquefying machine. In building it he took the help of the expansion turbine. Later he patented it. The expansion device was double flow one. Low temperature air entered at the centre and got divided into two opposite flowing streams. During that time, Joseph E. Johnson in USA created a device to liquefy the gas and patented it. His expander was a De Laval impulse turbine. A fraction of the 10 in-flowing air condensed in the turbine nozzle and then fell to the bottom of the liquefaction chamber for collection and run off.

Loads were applied on structural beams and shafts. Hence, the effect of crack on this type of structural element may affect the dynamic behaviour of the element and may be catastrophic. Any intentional change in the dimension of this type of structure may adversely affect its behaviour.

Linde Works, one of the largest gas exploration companies first used the expansion turbine successfully for the commercial application of liquefying the gas in 1930. That was an axial flow single stage expansion turbine. This was later replaced by an inward radial flow expansion

turbine. The new machine was invented by Guido Zerkowitz, an Italian engineer. In this new design, there was a reversing chamber fitted inside the turbine wheel. In this way by reducing the wheel speed, the velocity compounding could be achieved. Zerkowitz's patent gave many details of turbine construction to reduce refrigerative and piping losses. For instance, the shaft bearings were to be entirely outside the turbine housing, well removed from the cold zone.

In the year 1939, a Russian scientist Peter Kapitza[11], came up with two revolutionary conclusion in his paper. These two revolutionary conclusions are:

1. Basing his explanations on thermodynamic principles, he demonstrated that a low pressure (4-5 bar) liquefier using an expansion turbine would outperform a high pressure liquefier using a reciprocating expander. Besides having thermodynamic superiority, the low-pressure plant would be less expensive.
2. By both analytical and experimental studies, he demonstrated that an inward radial flow turbine would be preferable to an axial impulse type machine. His principal argument was that in an axial flow machine, the disruption of the regular flow by expansive air from the rotor would make it difficult to maintain radial equilibrium. The ejection was ascribed to radial pressure gradient generated by centrifugal forces in the rather dense working fluid. He advised that cryogenic turbines could follow the general design principles of radial inflow hydraulic machines, because both dealt with heavy fluids moving at comparatively low speeds.

During the World War II, standardized turbines were needed. Hydraulic machine concept was applied. Impeller parameters like specific speed ( $n_s$ ) and flow coefficient ( $\phi$ ) etc. became the

criteria for selection of turbines. The centrifugal inflow geometry thus became the usual configuration for small and medium sized cryogenic turbines [17].

In 1983 a Turboexpander was installed first in natural gas station by San Diego Gas and Electric. After that, additional installations were made in the mid 1980's in Memphis, Tennessee, Stockbridge, Georgia and Hamilton, New Jersey.

Turboexpander with secondary power generation source increase both overall output and efficiency.

In October of 2008, Enbridge opened a combination turbo expander and fuel cell facility in Toronto, Canada. The facility produces 2.2 megawatts.

## **2.2 - Recent Developments**

During the period of 1960s and 70s, demand of helium increased. Hence, the demand for energy efficient liquefiers and refrigerators based on turbines increased. The volumetric flow rate in case of helium turbines are low due to low fluid temperature and large pressure ratio. Because of that the operating speed of those turbines is very high, in the order of million revolutions per minute. At such high speed gas and magnetic bearings are the only alternatives.

Sixsmith first initiated research on small turbo-expanders in the period of 1905s at the University of Reading. The repercussion of the research came as: A prototype was successfully employed in a small laboratory sized air liquefier at the university. That brought a remarkable change. Later getting the idea of this small size turbo-expander, British Oxygen Company (BOC) manufactured expansion turbines for application in various air separation units. World's first viable turbine-based helium liquefier was built by BOC for the Rutherford Laboratory in Oxford. By the end 1950s, England based company "Lucas company" had developed a large number of gas lubricated inward flow turbines for PDC (Petrocarbon Development Corporation) [18].



Later many industries involved in the development process of the turbo-expander. Sulzer Bros., Switzerland, La Fleur Corporation, Lucas Corporation and Linde, Germany started constructing small turbines with gas as the lubrication medium for bearings.

By the 1980s, the design of turboexpanders with gas bearings as its support became almost popular in Europe and USA. They built two models of small turbines, one for helium liquefiers and another for small air separation units. Naka Fusion Research Centre affiliated to the Japan Atomic Energy Institute [13] modelled a very large size He turbo-expander. Ino et al. [14] developed a He turbo-expander for a 70 MW superconducting generator. The work of Kapitza, paved the way for the Russian Turbine industries to use both oil and gas bearings to support turbo-expander. This has continued through the 90s [10]. Small turboexpanders for microcryogenic systems have also been developed by the Mikrokryogenmash company in Russia[15].

Gas bearing technology, was applied to applications in small air separation plants. By considering stringent safety and reliability conditions, expansion turbine of Lucas was employed in a plant that was supplying blanket gas to a Uranium Hexafluoride (UF<sub>6</sub>) process. Later taking gas-bearing based turbine, scientists duo Sulzer Brothers developed a range of small nitrogen liquefiers (the LINIT series). Now a Germany company Linde AG, manufactures this capacity and range of laboratory liquefiers.

The development in the field like the micro turbines and application of bearings of a turboexpander are among various modern developments. A tiny version of the of the turbine for cryo-cooler was built by Sixsmith[16] in collaboration with Goddard Space Flight Centre of NASA. They built the turbine with diameter 1.5 mm and tested it with a speed of around one

million revolutions per minute. For cooling purpose like infrared detectors and electronic devices in space as well as ground, these devices are used. While gas bearings of both the aerostatic or aerodynamic varieties attained good amount of reliability, they still have process gas consumption, contact in starts and stops, sensitivity to dirt and shocks, and instabilities. Alternative bearing geometries were analysed by several authors. Foil bearings have been successfully used in cryogenic turbines. ACD Inc. in USA has used foil, as well as magnetic, bearings in their machines. Oil bearings have been replaced by magnetic bearings in many large scale applications including air separation plants and hydrocarbon industries.

### **2.3 - Organisations Involved in Turboexpander Activities**

Companies such as Air Products and Chemicals Inc., USA, L'Air Liquide of France and Linde AG of Germany manufacture turboexpanders. Creare Inc. [9], USA and Hitachi, Japan manufacture small helium turbines for small closed cycle cryorefrigerators based on the reverse Brayton cycle. Creare Inc. has built a range of highly compact and sophisticated turbines, the smallest of them having a wheel of only 1.5 mm diameter. Hitachi has produced small helium turbines for both gas phase and supercritical fluid flow. The Japan Atomic Energy Research Institute, in collaboration with Kobe Steel, manufactures large helium turbines for nuclear applications. Galiimash, Helium mash and Mikrokryogenmash are major cryogenic turboexpander manufacturers in Russia.

In India, after years and years of cryogenic research activity, the development in cryogenic expansion turbines has not been undertaken seriously. One exception is the work of Jadeja et [19]. al. at the Central Mechanical Engineering Research Institute, Durgapur. He developed a radial flow turbine with the support of gas bearings.

## **2.4 – Analysis of Turboexpander**

Turboexpander consist of following parts

1-shaft

2-bearings

3-thrust collar

4-expansion turbine

5-brake compressor

6-housings etc.

Generally various types of turbo machines are used for various types of application. Depending on size they can be categorized as small, medium and large turbo machines. Various works has been done by Linde AG of Germany, Creare Inc. [9] USA. Generally shaft, expansion turbine and brake compressor are taken for analysis of stress and deformation, as the whole assembly is kept in housing, there is a probability that due to high speed rotation of the Turboexpander, the various elements of it may grow radially decreasing the clearance between the housing and turbo expander. This may cause catastrophic damage to the housing and the turbo expander and also to the bearings. There are sharp edges present in shaft; brake compressor, turbine wheel, and those are major areas of stress concentration. As turboexpanders are very delicate and costly, it is imperative to analyze each and every part carefully before putting it into application.

### **Analysis of Turbomachine**

#### **3.1 - Objectives**

The objectives of the analysis are:

1. To find out the stress field in rotating components due to very high rotational speed and to carry out various modifications in the design to avoid failures of rotor.
2. To obtain the deformation of the various elements of the rotor, caused by the centrifugal force  
Centrifugal force caused by the high rotational speed deflects the material radially, decreasing the clearance between casing and the element. If the clearance decreases, then there is possibility of material, material contact causing wear and tear. That's why analysis of the deformation is inevitable.
3. To analyse the rotor system by varying the geometric feature like fillets, thickness on the shaft collar, compressor etc.

#### **3.2 - Steps for the Stress and Deflection Analysis**

Stress and deflection analysis have been carried out by using ANSYS fem solver.

1. First of all the models i.e the rotor, turbine and the compressor have been formulated by using parametric solid modelling software. We have used Solidworks 2012 for the modelling. The model is then saved in \*.iges/\*.igs format (iges stands for initial graphics exchange specification, which is a neutral data format file that helps in digital exchange of data among various CAD software).
2. This model is then imported to the ANSYS.

3. Meshing of the solid model is generated by using the meshing function of the ANSYS finite element solver package. Then various boundary conditions are applied to the model and solved for stress and deflection.

### 3.3 - Stress Analysis of Rotor Shaft

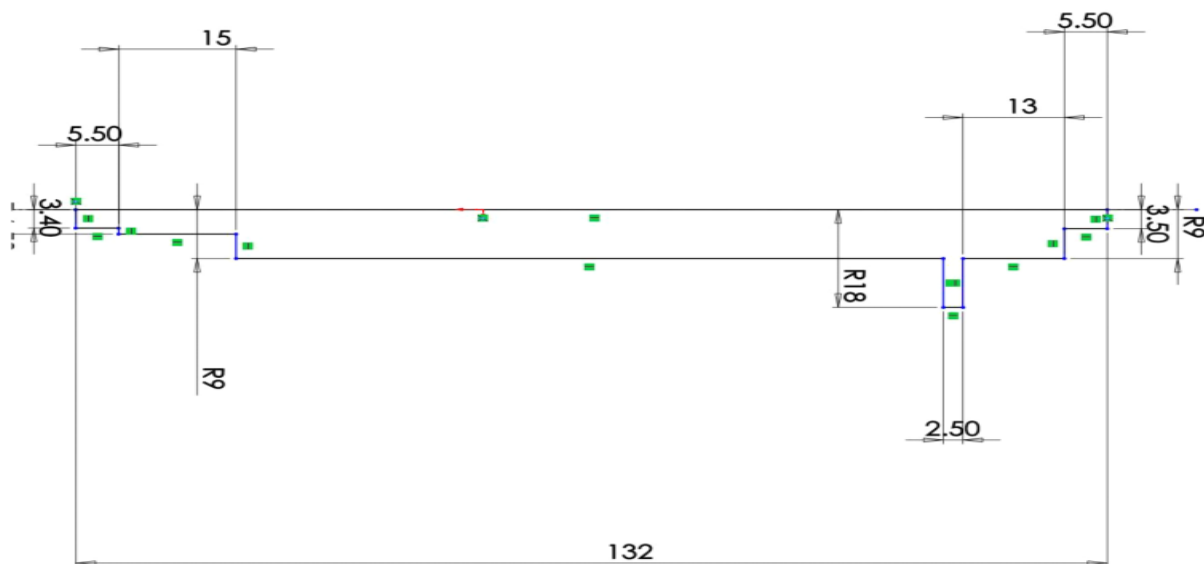
#### Objective

The shaft has been analysed to find out whether the maximum stress generated in rotor system is under the permissible limit or any modification is required to address the problem. Also the deformation analysis has been carried out to study the radial growth of the disc and the shaft at the bearing portion.

#### Solid Model and Mesh Generation

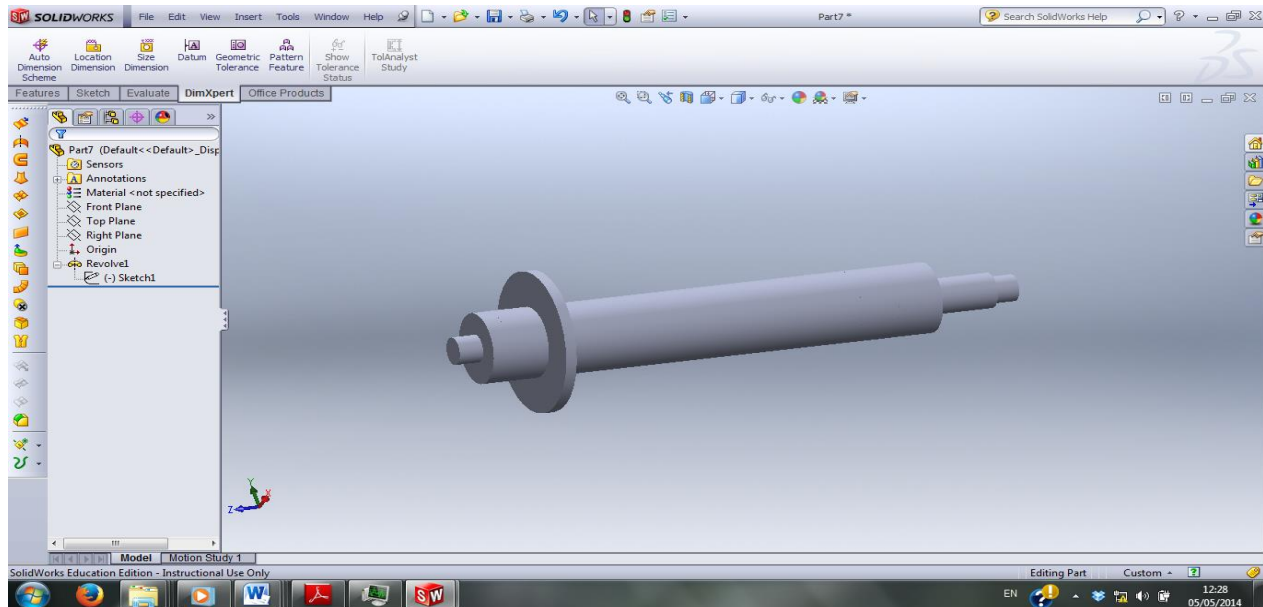
Processes used are as follows:

1. First a 2D drawing is prepared in the Solidworks 2012 software and as this geometry is symmetric it is revolved around an axis to create the solid model. This is then saved in \*.iges format.
2. The dimension of the shaft is shown in figure 3.1.



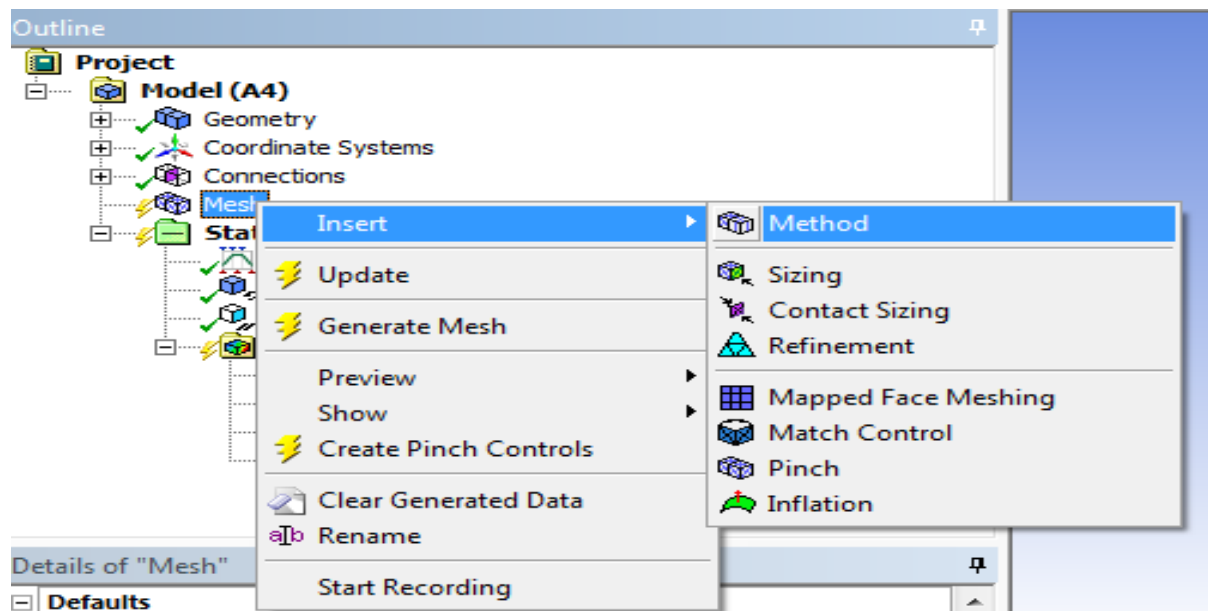
**Figure 3.1** –Dimension Of Rotor System (source – Ghosh P,(2002)[1]).

3. The figure 3.2 shows the model generation techniques in Solidworks 2012. The model is saved in iges format.



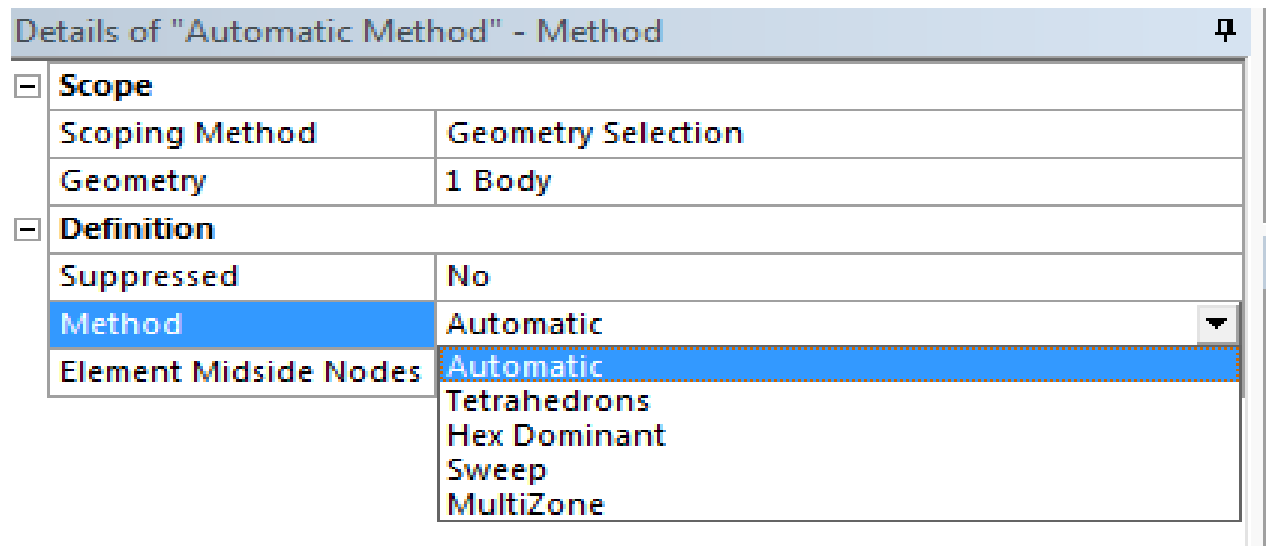
**Figure 3.2 – Solid Model of Rotor.**

4. The iges for the rotor is imported to ANSYS for the analysis.
5. Then mesh is generated using the mesh function of the ANSYS fem package. The figure 3.3 shows various processes and methods to create mesh.



**Figure 3.3 – Mesh Generation Process.**

The figure 3.4 shows various methods to create mesh.



**Figure 3.4** – Various Methods for Mesh Generation

Tetrahedral mesh generally provides more accurate solution in case of volume meshing than any other meshing methods that's why tetrahedron method has been chosen for meshing purpose.

### Boundary Conditions

A solid has three translational and three rotational degrees of freedom. To apply various boundary conditions to the model, some of these have to be constrained. By applying the bearing at the ends to rotational degrees of freedom is constrained i.e. the rotational motion along X-axis and Y-axis is zero as shaft axis is along Z direction. Then comes the displacement boundary condition. As the shaft does not have any lateral displacement, so the translational movement of the end points of the shaft along the X-axis and Y-axis are set to zero. Then rotational speed is applied to the shaft along Z-axis.

### Application of Load

The shaft is subjected to following loads:

1. Gravitational pull (g) is applied about axis of rotor(-Z Axis),as Rotor is vertically oriented.

2. Centrifugal force ( $m\omega^2 r$ ) in radially outward direction due to rotational speed.

## **Analysis and Results**

- 1) Mesh method – Tetrahedron; algorithm used – Patch conforming

- 2) Number of element- 34575

Number of nodes - 20403

- 3) Material used- structural steel having following properties

A. Density ( $\rho$ ) – 7850 kg/m<sup>3</sup>

B. Young's modulus (E) – 210 GPa

C. Poisson's ratio ( $\nu$ )– 0.3

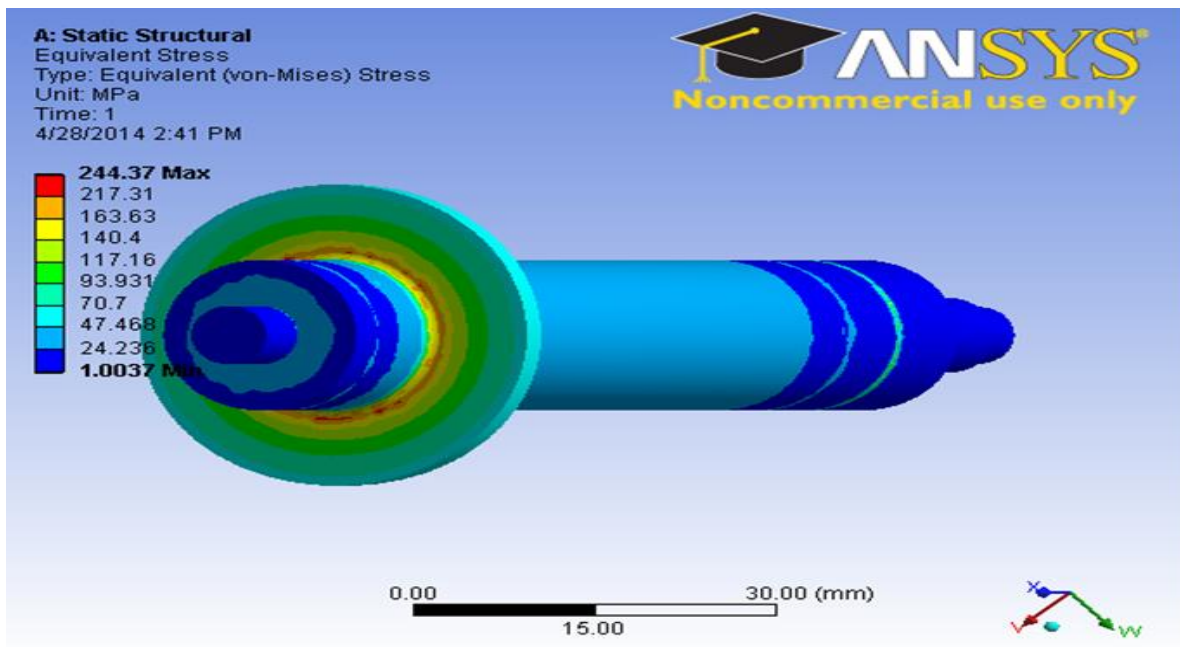
- 4) Rotational velocity applied is  $\omega = 14660$  rad/s.

- 5) After application of all boundary conditions, the model is solved for the stress and deformation analysis.

## **Stress analysis of shaft with Side Collar**

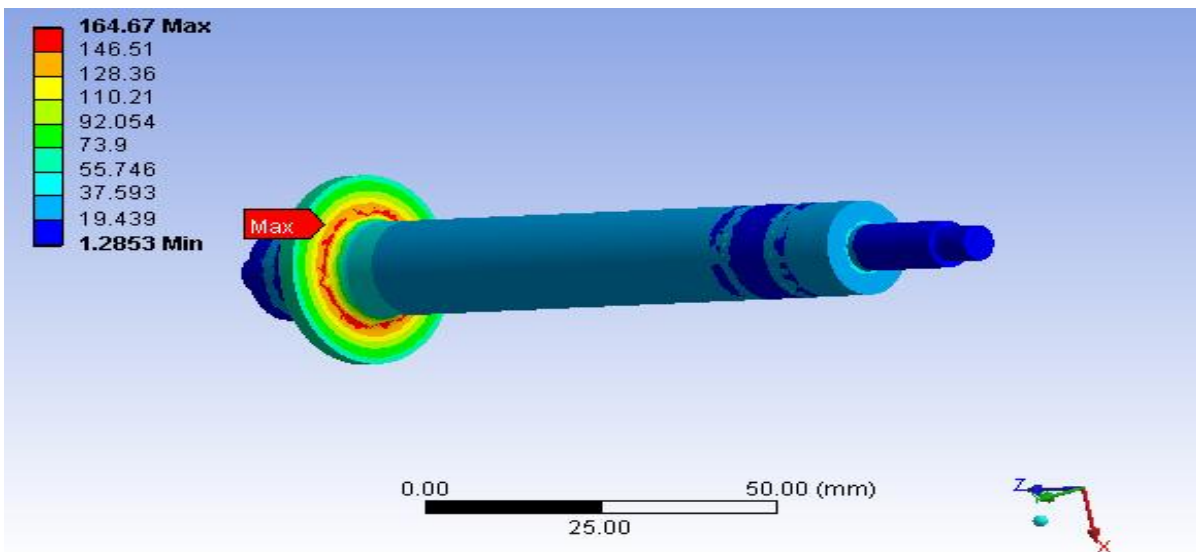
In small turbo machines, the position of collar is generally placed on one end of the rotor for providing space for sensors and instrumentation. The figure 3.5 shows the stress generated at different points of shaft. It can be seen from the figure that the stress at the root collar is 244MPa, which is very close to the yielding stress of the material considered in current analysis i.e. 250 MPa. That's why some modification in the design or the material is necessary. It is recommended to use some other high yield stress material other than structural steel. Before changing material of the shaft we can analysis by modifying the geometry with fillets at the critical areas. So for current analysis fillets are added to previous geometry at root of the collar with different radius.





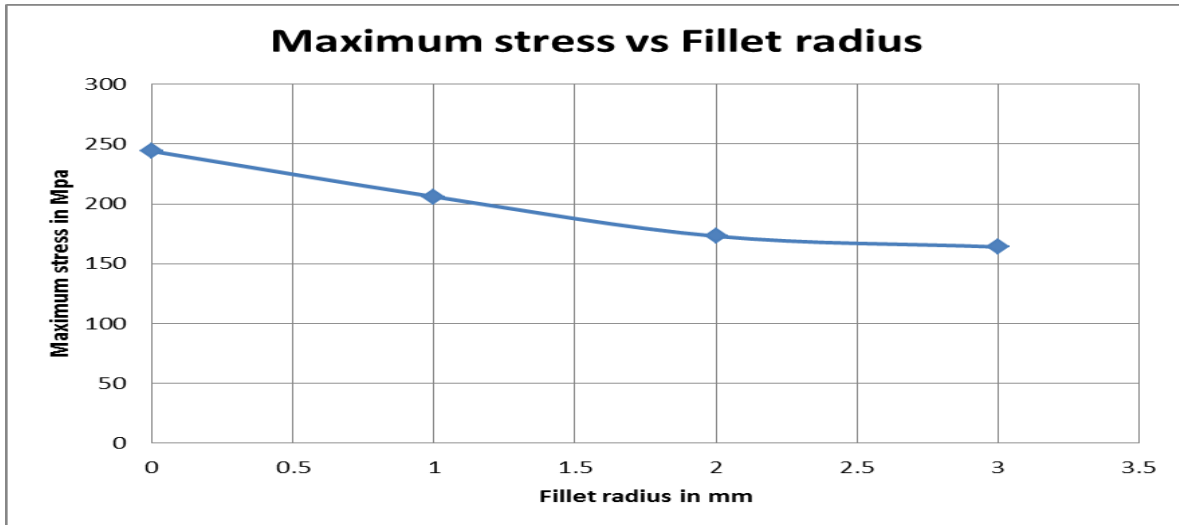
**Figure 3.5 – Von-Mises Stress of Shaft (Without Any Fillet) in MPa.**

Then fillet of 1mm, 2mm and 3mm is applied at the root of the shaft collar and then it's analysed. The figure 3.6 shows the stress pattern of the 3mm filleted shaft rotor system and it can be seen that the stress at the root of the disc is 164 MPa. By using fillet the stress decreased around 32% compared with the geometry without fillet.



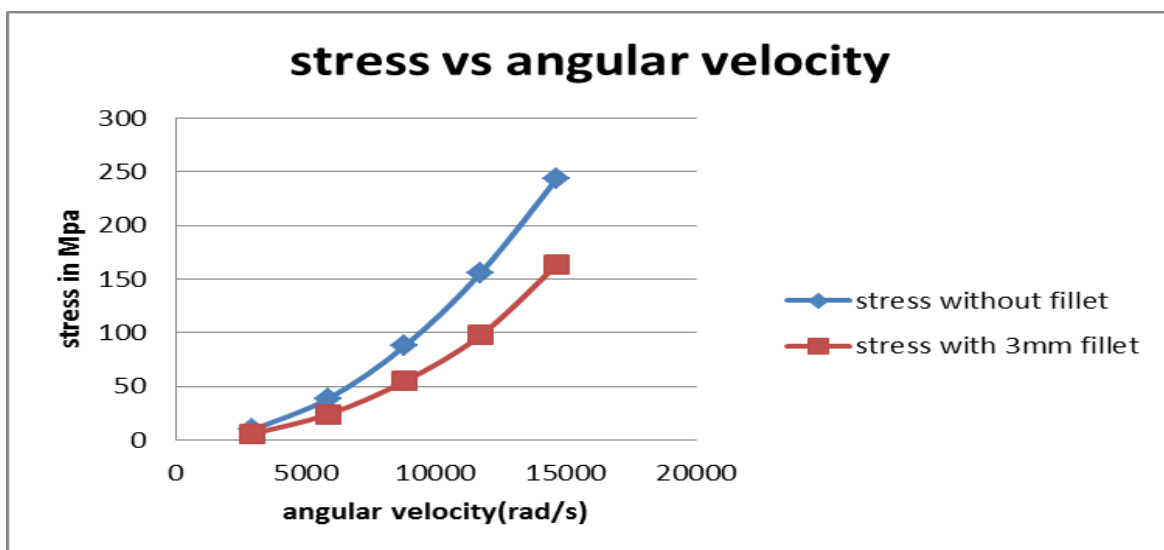
**Figure 3.6 – Von-Mises Stress on Shaft (with 3mm fillet) in MPa**

Similar analysis was done for fillet with radius 1 mm and 2 mm and their maximum stress on the rotor was found. The stress generated at the root portion of the modified designs versus different fillet radius is illustrated in the figure- 3.7.



**Figure 3.7** –Maximum Stress Vs Fillet Radius in rotor.

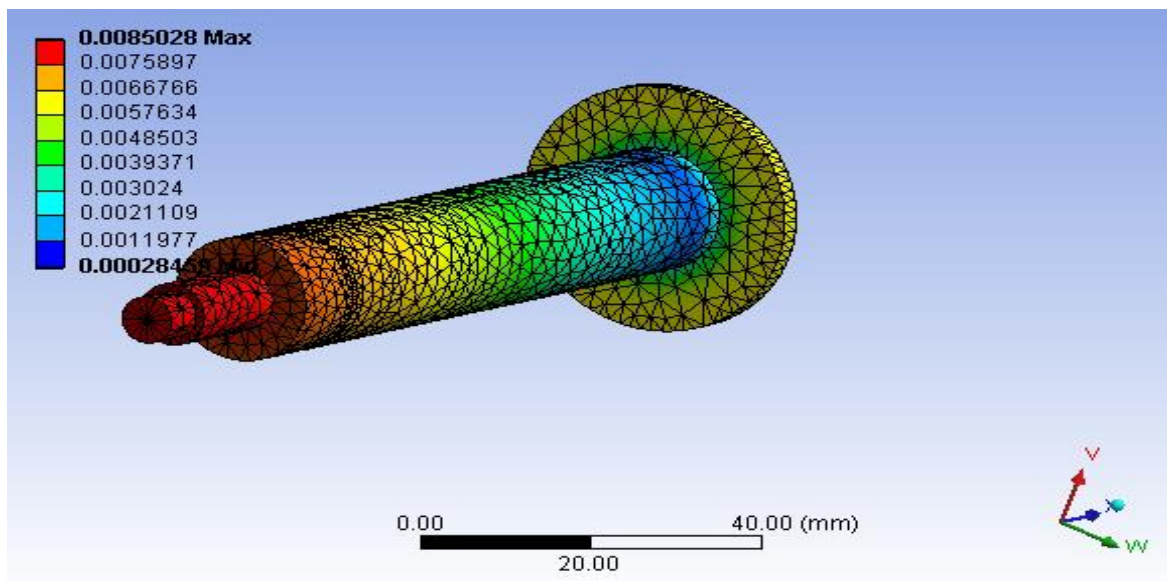
By taking the various values of stress at different rotational velocity for rotor with fillet and rotor without fillet radius of 3mm a chart has been prepared. The chart is shown in figure 3.8.



**Figure 3.8** – Variation of Stresses and Angular Velocity for Filleted and Non-Filleted Shaft Rotor

### Deformation analysis of shaft with Side Collar

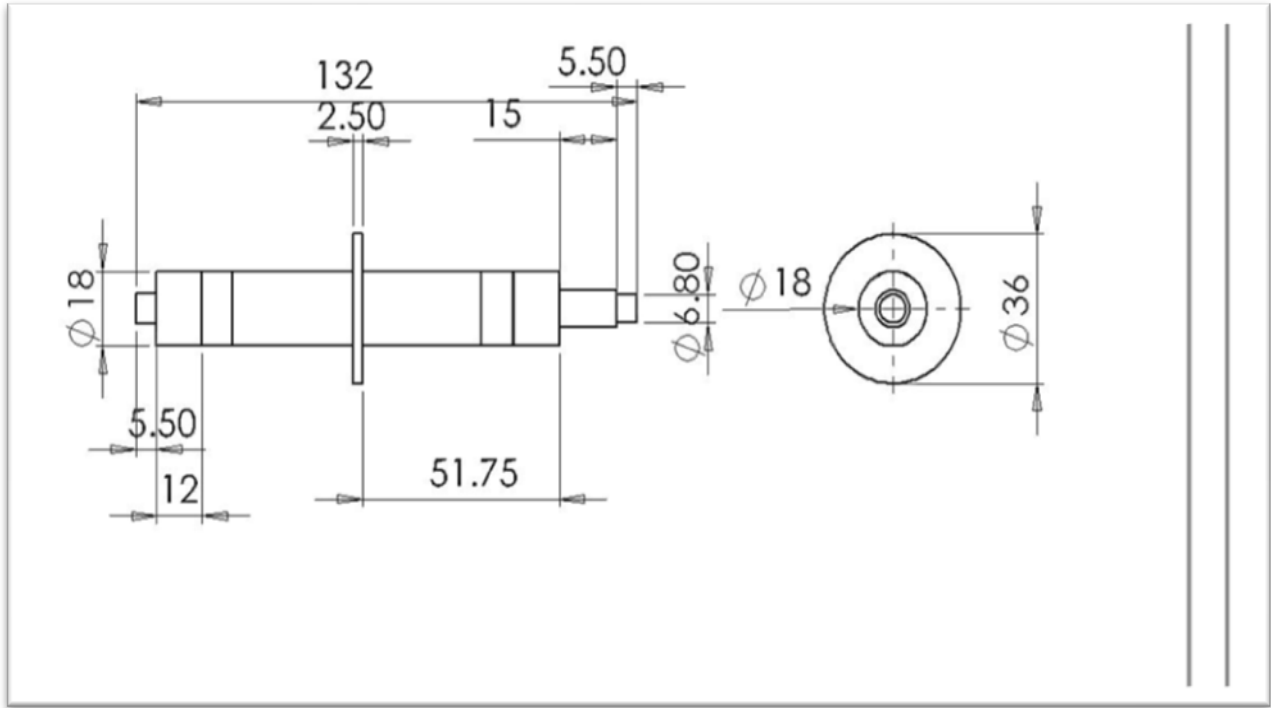
For better design of any rotor system both stress and deformation analysis is required. This section describes about the deformation analysis of rotor system used in current analysis. Figure 3.9 shows the deformation pattern of the rotor. It is found that one end of the rotor (opposite to the position of the collar) deformed to the maximum value of 0.008mm, which is a permissible for small/medium rotor system. However bearings placed at the end of the shaft, where the deformation is high may cause bearing failure. Next section describes about the stress and deformation analysis of shaft by shifting the collar to the middle of the shaft.



**Figure 3.9 – Displacement Plot of Shaft**

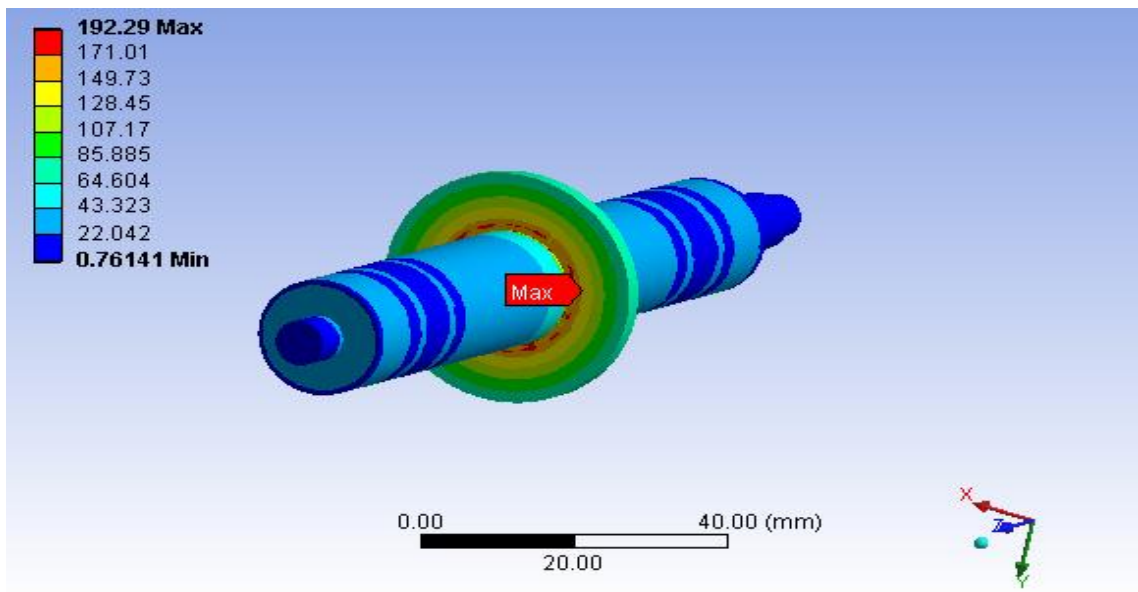
### Stress analysis of shaft with Middle Collar

The geometry of the shaft was changed by shifting the position of collar to middle of the shaft. The dimension of the new geometry is given in figure 3.10. For this modification both stress and deformation was analysed.

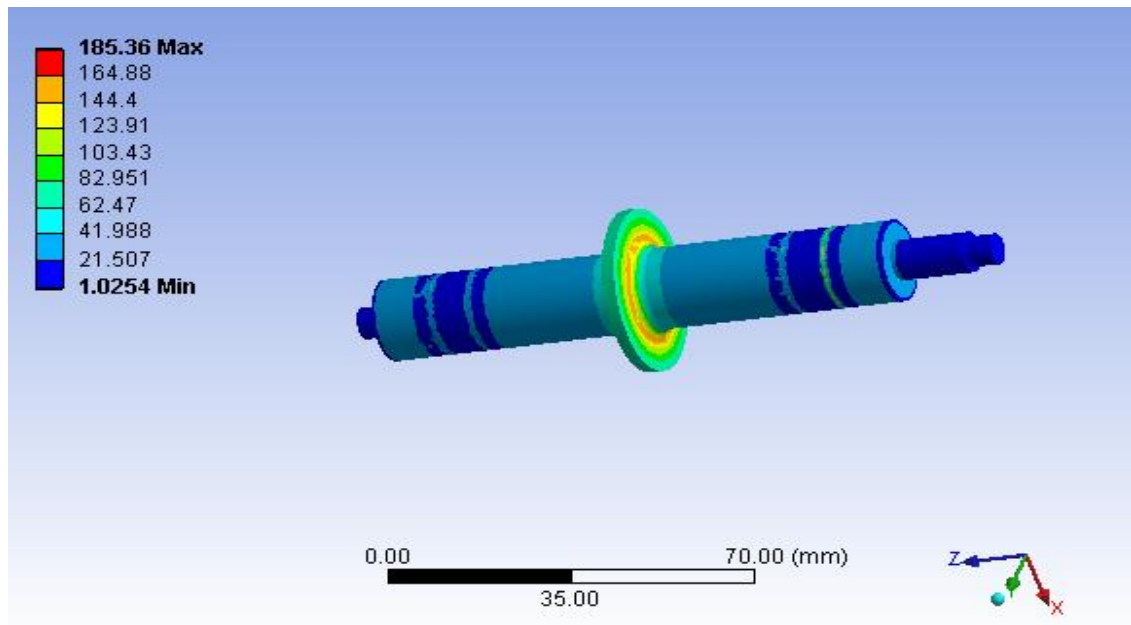


**Figure 3.10** – Dimension of the Rotor with Collar at Middle.

The figure 3.11 and figure 3.12 depicts the stress contour of the rotor with shaft without fillet and with fillet near root of the collar respectively. From the figure it was found that the maximum stress at the root of the disc is 192 MPa (without fillet) and 185Mpa (with fillet). This is much below the yield stress of the material. Also it was observed by shifting the collar to middle of the shaft, the maximum stress on the rotor was reduced by 21% compared with collar at one end of the shaft. The stress reduction was found to be about 24% less by using fillet, which is shown in the figure 3.12. As Most of the small high speed rotor moves at high speed and uses gas thrust bearing, the fillet near root of the collar may be an issue from flatness point of view.

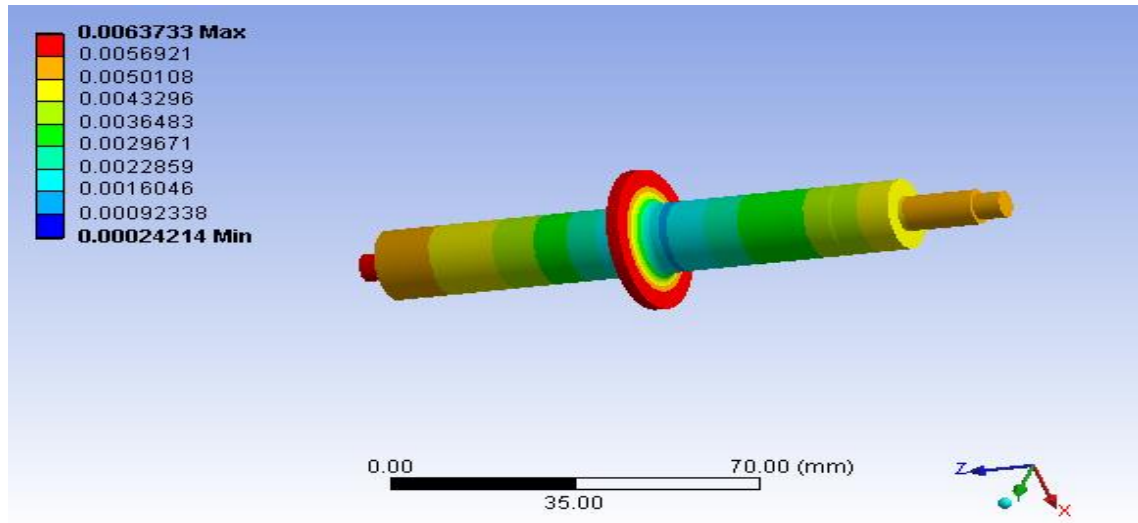


**Figure 3.11** – Von-Mises Stress of Shaft without fillet.



**Figure 3.12**– Von-Mises Stress of Shaft with fillet of radius 3 mm.

The deformation pattern of the rotor when it is situated at the middle of the shaft is shown below in the figure 3.13.

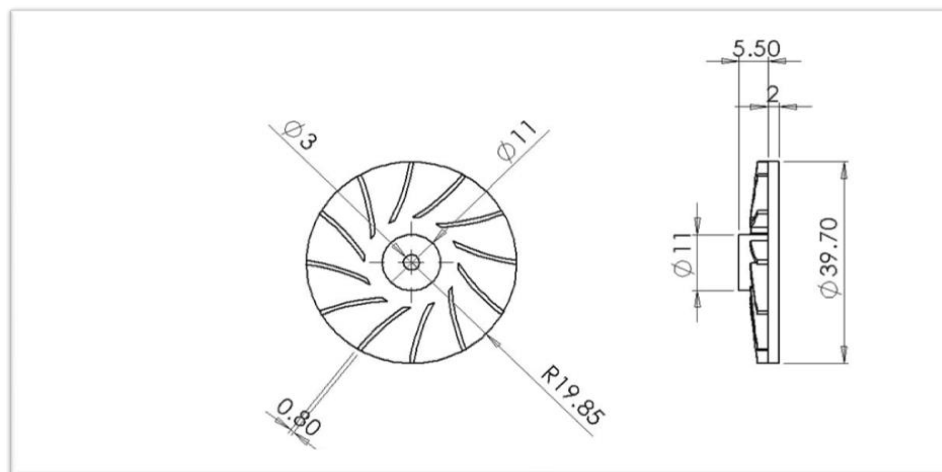


**Figure 3.13 – Deformation Pattern of the Shaft**

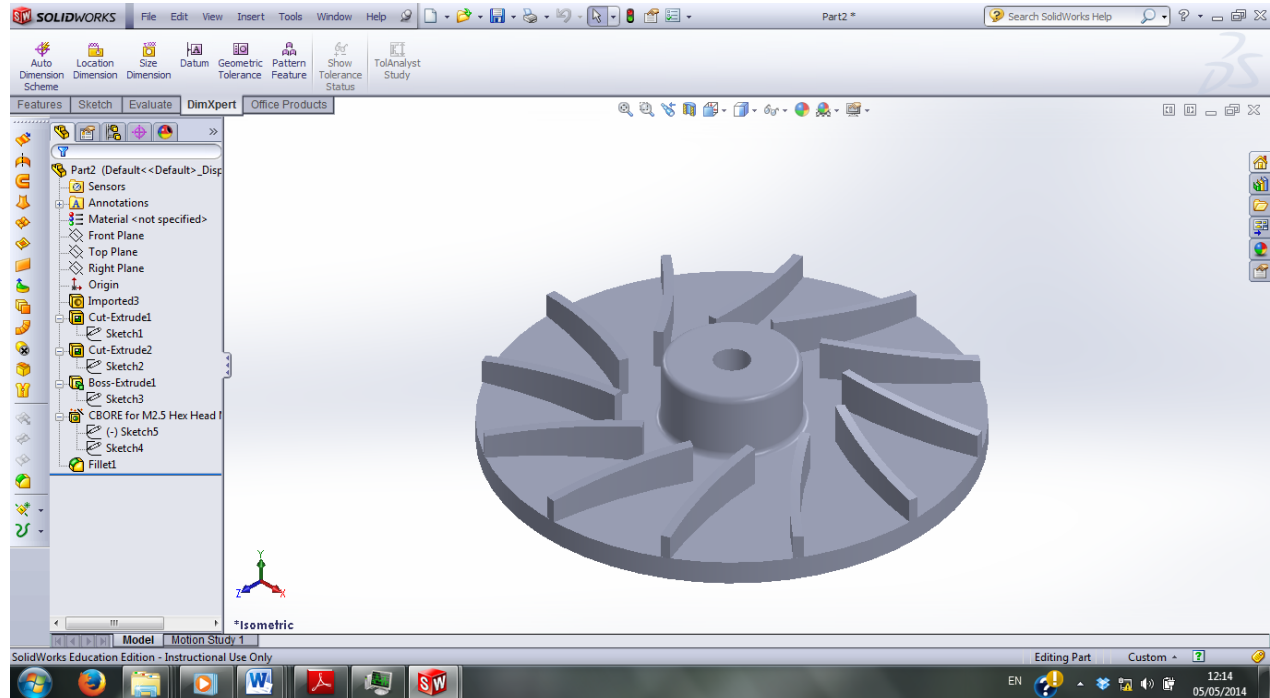
If we compare the deformation pattern of the rotor when the disc is situated at one side of the shaft with when it is at the middle, we can see that the radial growth of the disc is less when the disc is situated at the middle compared to when it is situated at one end.

### 3.4 – Compressor

Like shaft rotor system the compressor is also modelled in the CAD modelling software SolidWorks 2012 and imported to ANSYS fem solver package in the \*.iges format file. And the total process discussed in the shaft analysis is followed to get the stress and deformation. The dimension of the compressor and the model generation technique is given in the figure 3.14 and figure 3.15 respectively.



**Figure 3.14 – Dimension of the Compressor.**



**Figure 3.15 –3D Model of Compressor.**

### **Boundary Condition**

The compressor is applied to the fixed support at the hub to restrict its motion along any axis.

### **Load**

- (i) Centrifugal force is applied along Y-axis (Clockwise) to the rotor.
- (ii) Earth gravity is applied about Y-axis in negative direction.

### **Analysis and Results**

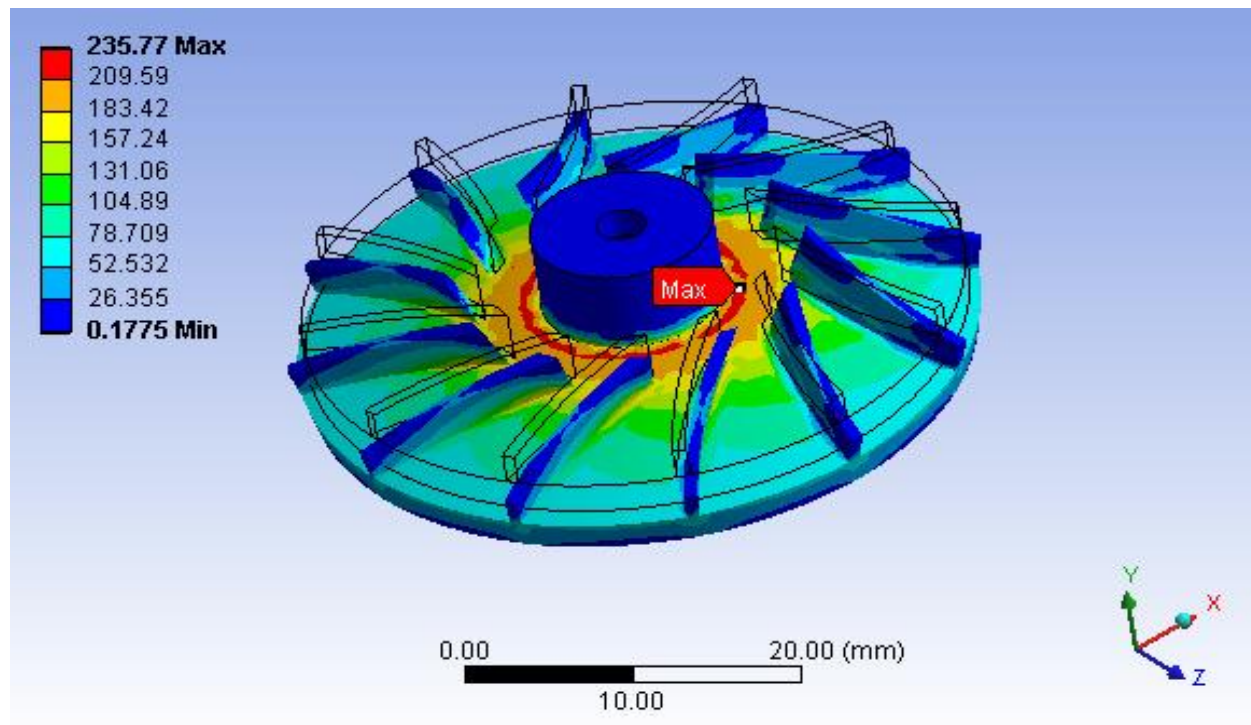
1. Element type -3D solid element
2. Number of element- 45354  
Number of nodes- 73442
3. Material used – Aluminium Alloy

- A. Density ( $\rho$ ):  $2700 \text{ kg/m}^3$
- B. Young's Modulus: 68.3 GPa
- C. Poisson's ratio: 0.33

4. Centrifugal load is applied to the compressor along Y-axis (Clockwise) having angular velocity  $\omega = 14660 \text{ rad/s}$ .
5. Earth gravity is applied about Y-axis,  $g = -9806.6 \text{ mm/s}^2$ .

### Stress Analysis

The figure 3.16 shows the stress contour of the compressor generated due to high rotational speed. It can be noticed that the maximum stress is generated near root of the compressor blade. i.e. 236 MPa, which is below the yielding stress of the aluminium alloy which is 276 MPa. So the compressor design is safe.



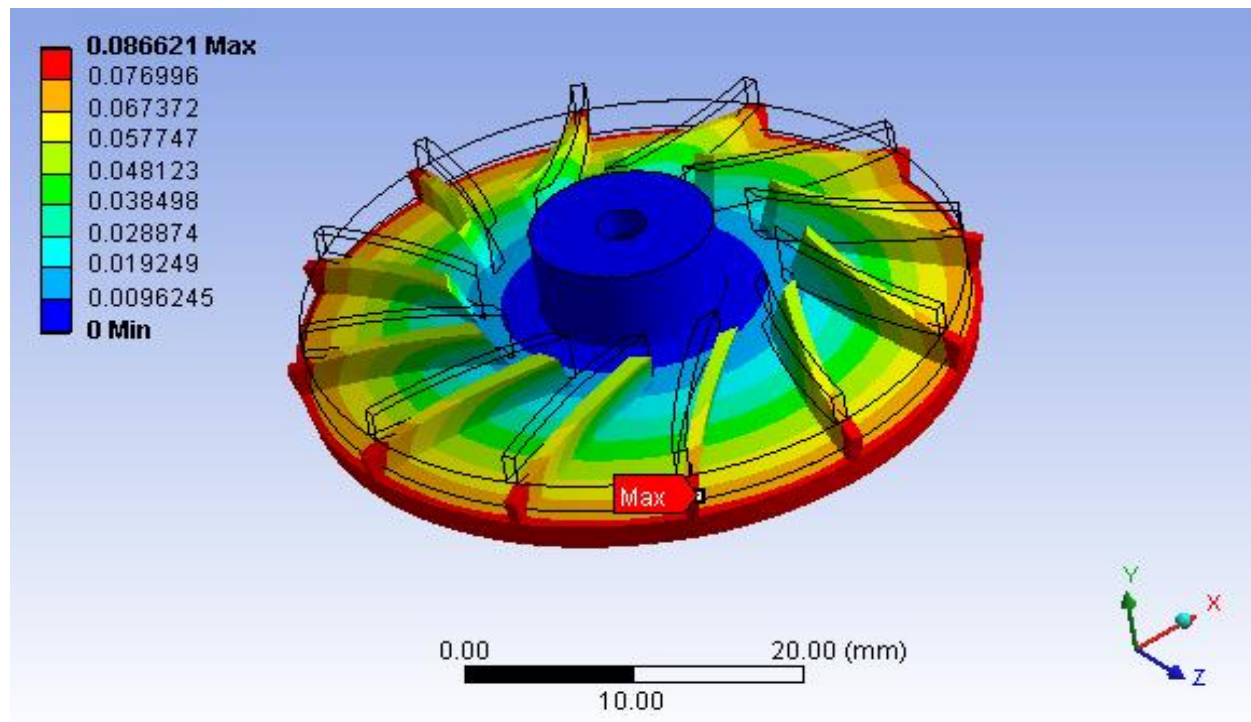
**Figure 3.16** – Von-Mises Stress Plot of Compressor.

### Deformation Analysis

Deformation analysis for compressor blade is very important as deformation of thin blade affects the efficiency of brake compressor. In current analysis figure 3.18 shows the deformation of the various elements of the compressor under high rotational velocity. From the figure 3.18 it can



be observed that the impeller blade bends downward under the application of the centrifugal force. The elements of the compressor try to be in line with the compressor axis under the application of the centrifugal force. The maximum deflection that is obtained from the analysis is 0.0867mm i.e the outer part of the compressor deformed more compare to the inner part. The red portion on the diagram shows the maximum deflection portion. As the radius of the impeller increases the deflection also increases. It shows that the deflection is proportional to the force applied.



**Figure 3.17 – Deformation Plot of Compressor.**

### **Analysis of Turbine Wheel**

Just like rotor and compressor the turbine wheel is shaped by SolidWorks 2012. As the blade of the turbine is of non-radial type, Table 1 shows coordinates of turbine blade at tip and hub, these coordinates help to generate blade profile using Solidworks.

### Tip Camberline

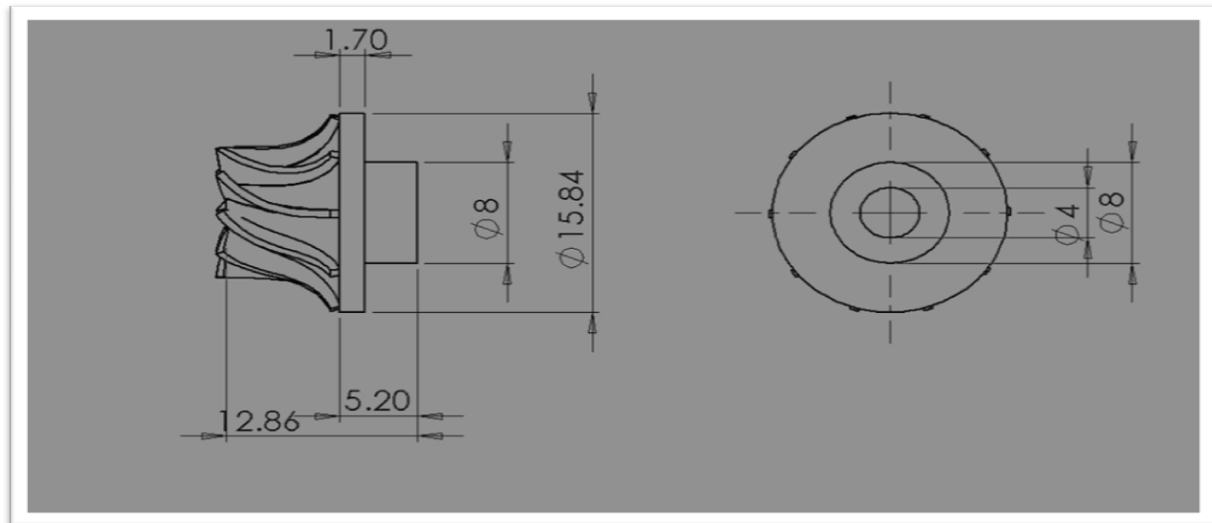
X	Y	Z
5.38	0	-0.24
5.253764936	0.618105164	0.24
5.098424852	1.120028582	0.71
4.958162484	1.53385944	1.18
4.828783121	1.874900951	1.63
4.719689274	2.158849961	2.08
4.635653487	2.399815983	2.52
4.579964924	2.607071402	2.95
4.545058067	2.78412413	3.37
4.53926371	2.943328893	3.79
4.562612444	3.08912086	4.19
4.614079317	3.226014888	4.58
4.699135081	3.365490973	4.97
4.809986091	3.502361176	5.34
4.959421486	3.653729399	5.69
5.139363507	3.814045456	6.02
5.355861304	3.992135982	6.33
5.599303915	4.184243739	6.62
5.869360949	4.390842977	6.87
6.164826432	4.613557744	7.09
6.485077156	4.853223081	7.28

### Hub Camberline

X	Y	Z
2.32	0	0.24
2.542464695	0.299120835	0.67
2.695718887	0.59219902	1.11
2.808670078	0.868891475	1.55
2.889812292	1.12204497	2
2.95548943	1.351880997	2.46
3.01939116	1.563098533	2.93
3.093866248	1.761133623	3.39
3.172160602	1.943141044	3.86
3.28068782	2.127248793	4.33
3.419889182	2.315439048	4.79
3.581443448	2.504029318	5.24
3.780446043	2.707531665	5.68
4.017753087	2.925501688	6.09
4.283136738	3.155493572	6.47
4.577245624	3.396884234	6.81
4.898849186	3.651489648	7.11
5.238833706	3.914871824	7.37
5.597112283	4.187174954	7.59
5.96466973	4.463766908	7.76
6.340964331	4.74537368	7.9

**Table 1** – Blade Profile of Turbine (Source – Ghosh S,(2008)[20]).

The dimension of the turbine wheel is given below in figure 3.18.



**Figure 3.18** – Dimension of Turbine Wheel.

After creating solid geometry, the body is imported to the ANSYS software by IGES file format. And meshing is done by the meshing function of ANSYS. Tetrahedron mesh generation method with element size 0.5 mm is used for meshing. Blade and hub interface has been refined by using the refinement option to get better result. After generating mesh, boundary conditions have been

applied to the geometry for analysis. Fixed support is applied to the hub and rotational speed of 14660 rad/s is given to the turbine around its axis.

### Analysis and Result

1. Material chosen- Aluminium Alloy

- a. Density ( $\rho$ ):  $2700 \text{ kg/m}^3$
- b. Young's Modulus: 68.3 GPa
- c. Poisson's ratio: 0.33

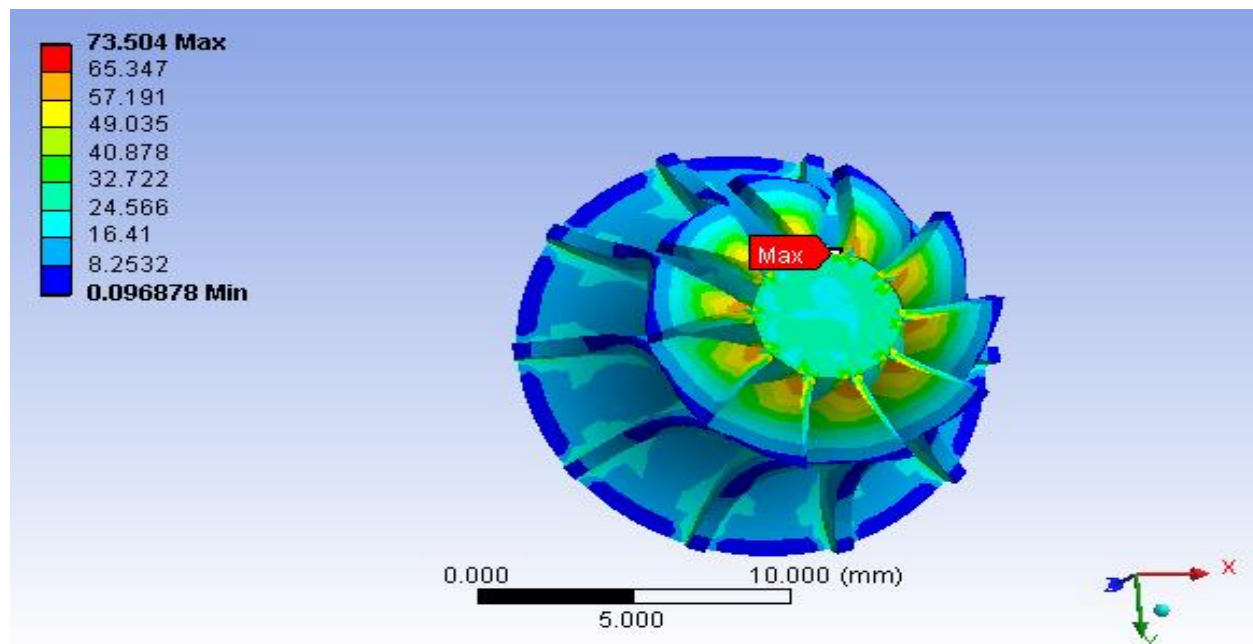
2. Number of element – 19591

Number of nodes - 5856

3. Load – centrifugal load with angular velocity 14660 rad/s along its axis.

#### Stress Analysis of expansion turbine

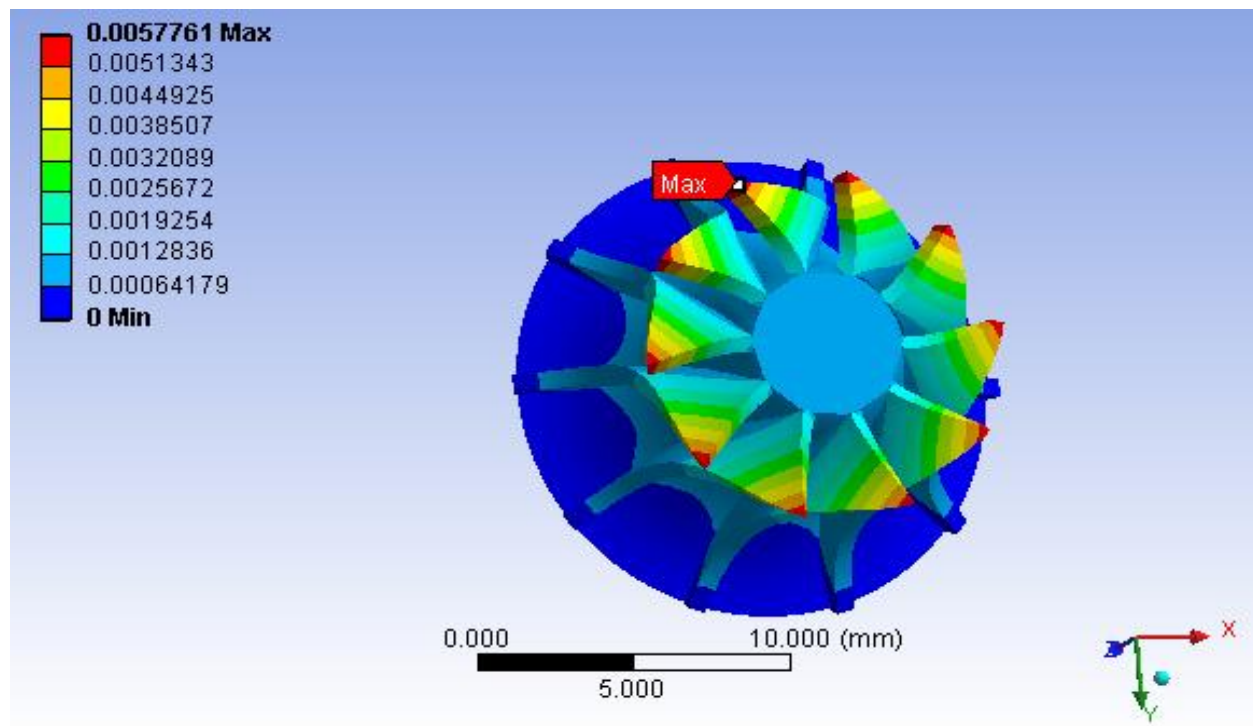
The figure 3.19 shows the stress distribution pattern of the turbine wheel. The stress obtained after analysis found to be very small i.e. 73 MPa. The low value of stress may be due to smaller size of turbine wheel compared to compressor. As the stress is very low than the yield stress of the aluminium alloy, the design is inherently safe.



**Figure 3.19-** Von-Mises Stress Plot of Turbine Wheel.

## Deformation Analysis

The figure 3.20 shows the deformation of the various elements of turbine wheel. It can be seen that the tip portion of the turbine wheel, which is showing red deformed more than any other parts. The maximum deformation is found to be within the permissible limit.



**Figure 3.20** – Deformed Shape of the Turbine Blades

### **Result and Discussion**

In the current investigation stress and deformation for different parts of turboexpander is done. Below are the lists of observation from current analysis.

1. Stress are found to be maximum in the root of the collar which are close to the yield stress of the material, so to avoid failure of rotor at high speed an attempt is made to redesign the rotor with different geometry by providing fillet of 3 mm at the root of the collar. By increasing the fillet size the stress can be further reduced but it may cause improper behavior of gas thrust bearing.
2. Further an attempt was made to reposition the location of shaft collar from compressor side end to the middle of the rotor. The analysis of such modification shows there is 21% reduction in maximum stress at the root of the collar. So current investigation suggest the position of the collar should be in the middle of the rotor.
3. The radial growth of the disc is more compared to any other part of the shaft. So the size of collar can be redesigned to reduce based on the load carrying capacity of thrust bearing.
4. The maximum stress distribution in the compressor was found to be near root of the blades. This can be further reduced by providing fillets at the root of the compressor blade. Similarly the deformation analysis shows to be high at the outer portion of compressor. To reduce the stress distribution and the deformation, author suggests changing the material of the compressor or redesigning the compressor blades.
5. The design of the turbine is most important in turboexpander, the current investigation found that the stress distribution of turbine wheel with radial entry and axial discharge designed based on Hasselgruber's technique is much below the yield stress of the material. Similarly the maximum deformation and deformation over the blades are found to be under permissible limit.

### **Conclusion and Future Scope**

Current project can be very helpful to the design engineer of turboexpander, which is used mostly for cryogenic application. The analysis will be very helpful to the engineering society of world to analyze, design and fabricate small turbomachinery using ANSYS.

Below is the list of future scopes for the project:

1. All components of turboexpander namely rotor, compressor and expansion turbine can be combined together and a detail analysis can be done to study the critical areas using ANSYS.
2. An analytical analysis of high speed rotor, compressor and expansion turbine using Finite Element Method can be done and compared with the result obtained from the ANSYS software.
3. Bearings for such high speed turbo machines are critical parts, so analysis of bearings are needed to complete design of turbo machines.

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